

Technical Note

Optimization of solar-powered Stirling heat engine with finite-time thermodynamics

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ABSTRACT

A mathematical model for the overall thermal efficiency of the solar-powered high temperature differential dish-Stirling engine with finite-rate heat transfer, regenerative heat losses, conductive thermal bridging losses and finite regeneration processes time is developed. The model takes into consideration the effect of the absorber temperature and the concentrating ratio on the thermal efficiency; radiation and convection heat transfer between the absorber and the working fluid as well as convection heat transfer between the heat sink and the working fluid. The results show that the optimized absorber temperature and concentrating ratio are at about 1100 K and 1300, respectively. The thermal efficiency at optimized condition is about 34%, which is not far away from the corresponding Carnot efficiency at about 50%. Hence, the present analysis provides a new theoretical guidance for designing dish collectors and operating the Stirling heat engine system.

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1. Introduction

Solar thermal power systems utilize the heat generated by a collector concentrating and absorbing the sun's energy to drive a heat engine/generator and produce electric power. Of the three solar thermal systems, the tower, the trough and the dish, the dish-Stirling systems have demonstrated the highest efficiency [1,16,19]. Over the last 20 years, eight different dish-Stirling systems ranging in size from 2 to 50 kW have been built by companies in the United States, Germany, Japan and Russia [1]. In principle, high concentrating and low or non-concentrating solar collectors can all be used to power the Stirling engine.

Chambadal and Novikov first applied finite-time thermodynamics to analyze heat engine in 1975, since then, extensive research have been undertaken on the performance analysis and optimization of Stirling heat engines and solar-powered Stirling engines based on FTT [2–20].

A part of the above publications are dedicated to the performance analysis and optimization of low temperature differential Stirling heat engines powered by low concentrating solar collectors [4,5,10,12,14,15,17]. In addition, finite-time thermodynamics analysis of heat engines is usually restricted to systems having either linear heat transfer law dependence to the temperature

differential both the reservoirs and engine working fluids [2–7,9,17,20]. However, for higher temperature solar-powered heat engines, radiation and convection modes of heat transfer are often coupled and play a collective role in the processes of engines [8]. Ahmet Z. Sahin [8] investigated the optimum operating conditions of endo-reversible heat engines with radiation and convection heat transfer between the heat source and working fluid as well as convection heat transfer between the heat sink and the working fluid based on simultaneous processes (according to [11], during the simultaneous processes, used as the steady-state operation in literature, the heat addition and heat rejection processes are assumed to take place simultaneously and are continuous in time as in a thermal power plant. The power output with simultaneous processes is given by $\dot{W} = (\dot{Q}_1 - \dot{Q}_2)$). Tamer Yilmaz et al. [13] investigated the optimum operating conditions of irreversible solar-powered Carnot-type heat engine with radiation dominated heat transfer between the heat source and the working fluid as well as convection dominated heat transfer between the heat sink and the working fluid based on simultaneous processes.

The objective of the present study is to investigate the optimal performance of internally and externally irreversible solar-powered high temperature differential dish-Stirling heat engines with radiation and convection heat transfer between the absorber and the working fluid as well as convection heat transfer between the heat sink and the working fluid based on sequential processes (according to [11], during the sequential or reciprocating processes, the four processes of the heat engine take place sequentially as in a

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Nomenclature

A	heat transfer area, m^2
C	collector concentrating ratio
C_v	specific heat capacity, $\text{J mol}^{-1} \text{K}^{-1}$
h	heat transfer coefficient, W K^{-1} or W K^{-4} or $\text{W m}^{-2} \text{K}^{-1}$
I	direct solar flux intensity, W m^{-2}
n	the mole number of the working fluid, mol
M	regenerative time constant, K s^{-1}
P	power, W
Q	heat transfer, J
q_u	heat gain, W
R	the gas constant, $\text{J mol}^{-1} \text{K}^{-1}$
t	time, s
T	temperature, K
W	Work, J
λ	ratio of volume during the regenerative processes
τ	cyclic period, s

η	thermal efficiency
δ	the Stefan's constant, $\text{W m}^{-2} \text{K}^{-4}$
k_0	heat leak coefficient, W K^{-1}
ε	emissivity factor
ε_R	effectiveness of the regenerator

Subscripts

app	collector
H	absorber
HC	high temperature side convection
HR	high temperature side radiation
L	heat sink
LC	low temperature side convection
m	the system
R	regenerator
t	Stirling engine
0	ambient or optics
1–4	the processes

reciprocating engine. The power output with sequential process is given by $\dot{W} = (\dot{Q}_1 t_1 - \dot{Q}_2 t_2) / \tau$. The influence of major parameters on the maximum power output and the corresponding overall efficiency is analyzed in detail. The aim of this article is to provide the basis for the design of a solar-powered high temperature differential Stirling engine operated with a high concentrating collector.

2. System description

The dish-Stirling system, comprising a parabolic dish collector (which is made up of a dish concentrator and a thermal absorber) and a Stirling heat engine located at the focus of the dish, tracks the sun and focuses solar energy into a cavity absorber where solar energy is absorbed and transferred to the Stirling engine to heat its displacer hot-end, thereby creating a solar-powered Stirling heat engine, as shown in Fig. 1.

Fig. 2 is a schematic diagram of a Stirling heat engine cycle with finite-time heat transfer and regenerative heat losses as well as conductive thermal bridging losses from the absorber to the heat sink. Fig. 3 is its T – V diagram. This cycle approximates the

compression stroke of real heat engine as an isothermal process 1–2, with an irreversible heat rejection at constant temperature T_2 to the heat sink at constant temperature T_L . The heat addition to the working fluid from the regenerator is modeled as isochoric process 2–3. The expansion stroke producing work is modeled as isothermal process 3–4, with irreversible heat addition at constant temperature T_1 from the absorber at constant temperature T_H . Finally, process 4–1 closed the cycle, the heat is rejected to the regenerator is modeled as isochoric process 4–1.

If the regenerator is ideal, the heat absorbed during process 4–1 should be equal to the heat rejected during process 2–3, however, the ideal regenerator requires an infinite area or infinite regeneration time to transfer finite heat amount, and this is impractical. Therefore, it is desirable to consider a real regenerator with heat losses ΔQ_R . In addition, we also consider conductive thermal bridging losses Q_0 from the absorber to the heat sink.

3. Thermodynamics analysis of the system

The analysis of the article includes the mathematical models for the dish solar collector, the Stirling engine as well as the combination of the dish solar collector and the Stirling engine. These models are described as follows.

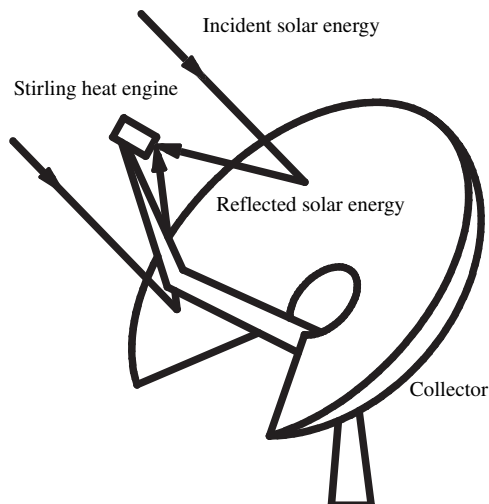


Fig. 1. Schematic diagram of the dish system.

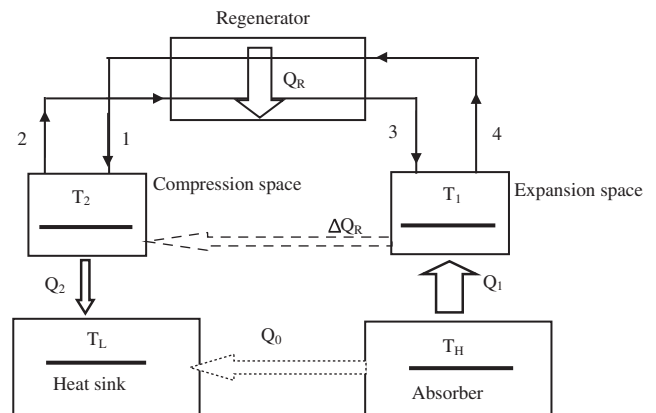


Fig. 2. Schematic diagram of the Stirling heat engine cycle.

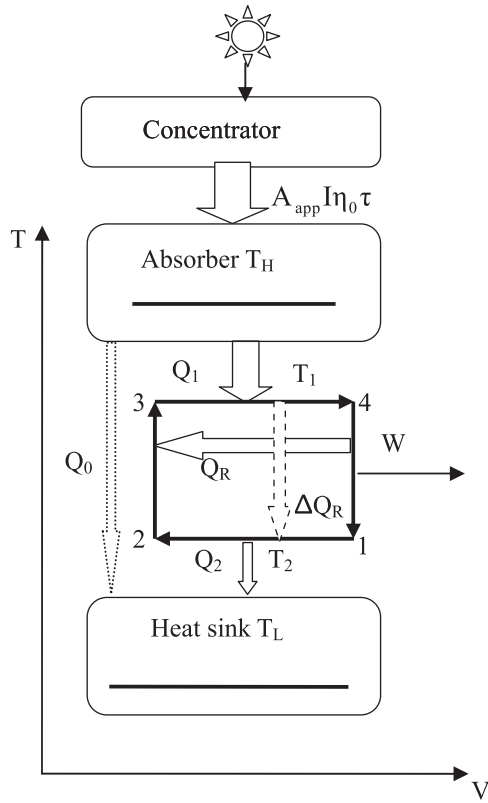


Fig. 3. T–V diagram of a solar Stirling engine cycle.

3.1. Thermal efficiency of the dish solar collector

Actual useful heat gain q_u of the dish collector, considering conduction, convection and radiation losses is given by [12,20]:

$$q_u = IA_{app}\eta_0 - A_{rec} \left[h(T_H - T_0) + \varepsilon\delta(T_H^4 - T_0^4) \right] \quad (1)$$

where I is the direct solar flux intensity, A_{app} is the collector aperture area, η_0 is the collector optical efficiency, A_{rec} is the absorber area, h is conduction/convection coefficient, T_H is the absorber temperature, T_0 is the ambient temperature, ε is emissivity factor of the collector, δ is the Stefan's constant.

Thermal efficiency η_s of the dish collector is:

$$\eta_s = \frac{q_u}{IA_{app}} = \eta_0 - \frac{1}{IC} \left[h(T_H - T_0) + \varepsilon\delta(T_H^4 - T_0^4) \right] \quad (2)$$

where C is the collector concentrating ratio.

3.2. Finite-time thermodynamics analysis of the Stirling heat engine

3.2.1. The conductive thermal bridging losses from the absorber to the heat sink

The conductive thermal bridging losses from the absorber at temperature T_H to the heat sink at temperature T_L is assumed to be proportional to the cycle time and given by [7,11]:

$$Q_0 = k_0(T_H - T_L)\tau \quad (3)$$

where k_0 is the heat leak coefficient between the absorber and the heat sink, τ is the cyclic period.

3.2.2. Regenerative heat losses of the regenerator

Let ΔQ_R be the heat losses during two regenerative processes per cycle and given by [5–7]:

$$\Delta Q_R = nC_v(1 - \varepsilon_R)(T_1 - T_2) \quad (4)$$

Then, the regenerative heat transfer Q_R is:

$$Q_R = nC_v\varepsilon_R(T_1 - T_2) \quad (5)$$

where n is the mole number of the working fluid, C_v is the specific heat capacity of the working fluid per mole in the regenerative processes, ε_R is the effectiveness of the regenerator, T_1 and T_2 are temperatures of the working fluid in the high temperature isothermal process 3–4 and in the low temperature isothermal process 1–2, respectively.

Owing to the influence of irreversibility of the finite-rate heat transfer, the time of the regenerative processes is not negligible in comparison to that of the two isothermal processes [7]. In order to calculate the time of the regenerative processes, one assumes that the temperature of the working fluid in the regenerative processes as a function of time is given by [20]:

$$\frac{dT}{dt} = \pm M_i \quad (6)$$

where M is the proportionality constant which is independent of the temperature difference and dependent only on the property of the regenerative material, called regenerative time constant and the $\pm$ sign belong to the heating ($i = 1$) and cooling ($i = 2$) processes respectively.

One obtains the time of the two isochoric processes as:

$$t_3 = \frac{T_1 - T_2}{M_1} \quad (7)$$

$$t_4 = \frac{T_1 - T_2}{M_2} \quad (8)$$

3.2.3. The amounts of heat released by the absorber and absorbed by the heat sink

On the one hand, we consider that radiation and convection modes of heat transfer between the absorber and the working fluid. Let Q_1 be the amount of heat absorbed by the working fluid at temperature T_1 from the absorber at temperature T_H .

$$\begin{aligned} Q_1 &= \left[h_{HC}(T_H - T_1) + h_{HR}(T_H^4 - T_1^4) \right] t_1 \\ &= nRT_1 \ln \lambda + nC_v(1 - \varepsilon_R)(T_1 - T_2) \end{aligned} \quad (9)$$

On the other hand, convection heat transfer is assumed to be the main mode of heat transfer between the heat sink and the working fluid. Let Q_2 be the amount of heat released by working fluid at temperature T_2 to the heat sink at temperature T_L .

$$Q_2 = h_{LC}(T_2 - T_L)t_2 = nRT_2 \ln \lambda + nC_v(1 - \varepsilon_R)(T_1 - T_2) \quad (10)$$

where h_{HC} is high temperature side convection heat transfer coefficient, h_{HR} is high temperature side radiation heat transfer coefficient, h_{LC} is low temperature side convection heat transfer coefficient, R is the gas constant, and λ is the ratio of volume during the regenerative processes, i.e.

$$\lambda = \frac{V_1}{V_2} \quad (11)$$

Take into account the major irreversibility mentioned above, the net heats released from the absorber Q_H and absorbed by the heat sink Q_L are given as:

$$Q_H = Q_1 + Q_0 \quad (12)$$

$$Q_L = Q_2 + Q_0 \quad (13)$$

3.2.4. The cyclic period

Using Eqs. (7)–(10), we get that the cyclic period τ is:

$$\eta_t = \frac{T_1 - xT_1}{T_1 + A_1(T_1 - xT_1) + [k_0(T_H - T_L)] \left[\frac{T_1 + A_1(T_1 - xT_1)}{h_{HC}(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \frac{xT_1 + A_1(T_1 - xT_1)}{h_{LC}(xT_1 - T_L)} + F_1(T_1 - xT_1) \right]} \quad (20)$$

$$\begin{aligned} \tau &= t_1 + t_2 + t_3 + t_4 \\ &= \frac{nRT_1 \ln \lambda + nC_v(1 - \varepsilon_R)(T_1 - T_2)}{h_{HC}(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \frac{nRT_2 \ln \lambda + nC_v(1 - \varepsilon_R)(T_1 - T_2)}{h_{LC}(T_2 - T_L)} \\ &\quad + \left(\frac{1}{M_1} + \frac{1}{M_2} \right) (T_1 - T_2) \end{aligned} \quad (14)$$

3.2.5. Maximum power output and maximum power efficiency of the Stirling engine

The power output and the thermal efficiency are given by [20]:

$$P = \frac{W}{\tau} = \frac{Q_H - Q_C}{\tau} \quad (15)$$

$$\eta_t = \frac{Q_H - Q_C}{Q_H} \quad (16)$$

$$P_{\max} = \frac{1 - x}{\frac{1 + A_1(1 - x)}{h_{HC}(T_H - T_{1\text{opt}}) + h_{HR}(T_H^4 - T_{1\text{opt}}^4)} + \frac{x + A_1(1 - x)}{h_{LC}(xT_{1\text{opt}} - T_L)} + F_1(1 - x)} \quad (22)$$

$$\eta_{t\text{opt}} = \frac{1 - x}{1 + A_1(1 - x) + [k_0(T_H - T_L)] \left[\frac{1 + A_1(1 - x)}{h_{HC}(T_H - T_{1\text{opt}}) + h_{HR}(T_H^4 - T_{1\text{opt}}^4)} + \frac{x + A_1(1 - x)}{h_{LC}(xT_{1\text{opt}} - T_L)} + F_1(1 - x) \right]} \quad (23)$$

Using Eqs. (9)–(16), we have:

$$P = \frac{T_1 - T_2}{\frac{T_1 + A_1(T_1 - T_2)}{h_{HC}(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \frac{T_2 + A_1(T_1 - T_2)}{h_{LC}(T_2 - T_L)} + F_1(T_1 - T_2)} \quad (17)$$

$$\eta_t = \frac{T_1 - T_2}{T_1 + A_1(T_1 - T_2) + [k_0(T_H - T_L)] \left[\frac{T_1 + A_1(T_1 - T_2)}{h_{HC}(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \frac{T_2 + A_1(T_1 - T_2)}{h_{LC}(T_2 - T_L)} + F_1(T_1 - T_2) \right]} \quad (18)$$

where

$$A_1 = \frac{C_v(1 - \varepsilon_R)}{R \ln \lambda}$$

and

$$F_1 = \frac{1}{nR \ln \lambda} \left(\frac{1}{M_1} + \frac{1}{M_2} \right)$$

For the sake of convenience, a new parameter $x = T_2/T_1$ is introduced into Eqs. (17) and (18), then we have:

$$P = \frac{T_1 - xT_1}{\frac{T_1 + A_1(T_1 - xT_1)}{h_{HC}(T_H - T_1) + h_{HR}(T_H^4 - T_1^4)} + \frac{xT_1 + A_1(T_1 - xT_1)}{h_{LC}(xT_1 - T_L)} + F_1(T_1 - xT_1)} \quad (19)$$

To maximize the power output, take the derivative of the Eq. (19) with respect to the temperature T_1 and equate it to zero, namely $\partial P / \partial T_1 = 0$, the optimal working fluid temperature $T_{1\text{opt}}$ for this condition can be obtained from Eq. (21):

$$D_1 T_{1\text{opt}}^8 + D_2 T_{1\text{opt}}^5 + D_3 T_{1\text{opt}}^4 + D_4 T_{1\text{opt}}^3 + D_5 T_{1\text{opt}}^2 + D_6 T_{1\text{opt}} + D_7 = 0 \quad (21)$$

where $D_1 = h_{HR}^2 B_1 x$, $D_2 = h_{HR} x (2B_1 h_{HC} - 3B_2 h_{LC} x)$, $D_3 = 2h_{HR} x (3B_2 h_{LC} T_L - B_1 B_3)$, $D_4 = -3B_2 h_{HR} h_{LC} T_L^2$, $D_5 = h_{HC} x (B_1 h_{HC} - B_2 h_{LC} x)$, $D_6 = 2h_{HC} x (B_2 h_{LC} T_L - B_3 B_1)$, $D_7 = B_1 x B_3^2 - B_2 h_{HC} h_{LC} T_L^2$, $B_1 = x + A_1(1 - x)$, $B_2 = 1 + A_1(1 - x)$, $B_3 = h_{HC} T_H + h_{HR} T_H^4$.

Therefore, the maximum power output and the corresponding optimal thermal efficiency of the Stirling engine are:

Special cases:

- (1) when $h_{HR} = 0$, i.e. only convection heat transfer is considered and radiation heat transfer is neglected between the absorber and the working fluid, Eq. (21) is simplified to:

$$D_5 T_{1\text{opt}}^2 + D_6 T_{1\text{opt}} + D_7 = 0 \quad (24)$$

Solve the Eq. (24), we get:

$$T_{1\text{opt}} = \frac{T_L + G_1 T_H}{x + G_1} \quad (25)$$

where

$$G_1 = \left[\frac{x^2 + A_1 x(1-x)}{1 + A_1(1-x)} \right]^{0.5}$$

The Eq. (25) is the same as the formula 4-46 given by [20].

(2) when $h_{HR} = 0$, if $\varepsilon_R = 1$, i.e. the Stirling engine reaches the condition of ideal regeneration, let the $x = \sqrt{T_L/T_H}$, we have:

$$P_{\max} = \frac{G_2 (\sqrt{T_H} - \sqrt{T_L})^2}{[1 + G_2 F_1 (\sqrt{T_H} - \sqrt{T_L})^2]} \quad (26)$$

$$\eta_{\text{topt}} = 1 - \sqrt{\frac{T_L}{T_H}} \quad (27)$$

where

$$G_2 = \frac{h_{HC} h_{LC}}{(\sqrt{h_{HC}} + \sqrt{h_{LC}})^2}$$

$$\eta_m = \left\{ \eta_0 - \frac{1}{IC} [h(T_H - T_0) + \epsilon \delta (T_H^4 - T_0^4)] \right\} \frac{1-x}{1 + A_1(1-x) + [k_0(T_H - T_L)] \left[\frac{1 + A_1(1-x)}{h_{HC}(T_H - T_{\text{topt}}) + h_{HR}(T_H^4 - T_{\text{topt}}^4)} + \frac{x + A_1(1-x)}{h_{LC}(xT_{\text{topt}} - T_L)} + F_1(1-x) \right]} \quad (29)$$

The Eqs. (26) and (27) are the same as the formulas (24) and (25) given by [6].

Performing numerical calculation for Eq. (21), we get Table 1.

From Table 1, we know that the temperature differential between the absorber and the working fluid increases with the increasing of the absorber temperature, therefore, for a higher temperature Stirling heat engine, radiation heat transfer between the absorber and the working fluid should not be neglected, we should consider both radiation and convection heat transfer between the absorber and the working fluid. It is also found that for a given absorber temperature, the optimal temperature of the working fluid decreases with the increasing of the effectiveness of the regenerator.

3.3. The maximum power thermal efficiency of the system

The maximum power thermal efficiency of the system is product of the thermal efficiency of the collector and the optimal thermal efficiency of the Stirling engine [12]. Namely:

$$\eta_m = \eta_s \eta_{\text{topt}} \quad (28)$$

Using Eqs. (2) and (23), we get:

Table 1
 T_{topt} for different T_H and ε_R .

T_H	ε_R												
	0.40	0.45	0.50	0.55	0.60	0.65	0.70	0.75	0.80	0.85	0.90	0.95	1.0
650	645.81	645.8	645.75	645.74	645.72	645.7	645.65	645.61	645.58	645.53	645.46	645.39	645.31
700	675.12	675.02	674.9	674.76	674.62	674.45	674.26	674.04	673.79	673.49	673.15	672.72	672.19
750	705.1	704.91	704.69	704.45	704.18	703.87	703.53	703.13	702.67	702.12	701.49	700.7	699.74
800	735.81	735.53	735.22	734.87	734.48	734.03	733.53	732.96	732.29	731.5	730.57	729.44	728.03
850	767.33	766.96	766.55	766.09	765.58	765.28	764.34	763.59	762.71	761.69	760.47	758.98	757.15
900	799.69	799.24	798.73	798.17	797.54	796.83	796.02	795.09	794.01	792.75	791.24	789.41	787.14
950	832.96	832.42	831.82	831.15	830.4	829.56	828.6	827.5	826.23	824.72	822.94	820.77	818.07
1000	867.15	866.53	865.84	865.07	864.21	863.24	862.13	860.87	859.39	857.66	855.6	853.09	849.98
1050	902.3	901.6	900.82	899.95	898.98	897.88	896.63	895.2	893.53	891.57	889.24	886.4	882.88
1100	938.4	937.62	936.75	935.79	934.71	933.49	932.1	930.51	928.66	926.48	923.88	920.73	916.8
1150	975.44	974.59	973.64	972.58	971.4	970.06	968.54	966.79	964.76	962.37	959.52	956.05	951.73
1200	1013.4	1012.5	1011.5	1010.3	1009	1007.6	1005.9	1004	1001.8	999.23	996.13	992.36	987.66
1250	1052.3	1051.3	1050.2	1049	1047.6	1046	1044.2	1042.2	1039.8	1037	1033.7	1029.6	1024.6
1300	1092	1090.9	1089.8	1088.5	1087	1085.3	1083.4	1081.3	1078.7	1075.7	1072.2	1067.8	1062.4
1350	1132.6	1131.4	1130.2	1128.8	1127.2	1125.5	1123.5	1121.2	1118.5	1115.3	1111.6	1106.9	1101.2
1400	1173.9	1172.7	1171.4	1169.9	1168.3	1166.4	1164.3	1161.9	1159.1	1155.7	1151.8	1146.9	1140.8
1450	1215.9	1214.7	1213.3	1211.8	1210.1	1208.1	1205.9	1203.4	1200.4	1196.9	1192.7	1187.6	1181.3
1500	1258.6	1257.3	1255.9	1254.3	1252.5	1250.5	1248.2	1245.5	1242.5	1238.8	1234.5	1229.1	1222.5
1550	1302	1300.6	1299.1	1297.5	1295.6	1293.5	1291.1	1288.4	1285.2	1281.4	1276.9	1271.3	1264.4
1600	1345.9	1344.5	1343	1341.3	1339.3	1337.2	1334.7	1331.8	1328.5	1324.6	1319.9	1314.1	1307
1650	1390.3	1388.9	1387.3	1385.6	1383.6	1381.3	1378.8	1375.8	1372.4	1368.4	1363.5	1357.6	1350.2
1700	1435.3	1433.8	1432.2	1430.4	1428.3	1426	1423.4	1420.4	1416.8	1412.7	1407.7	1401.6	1393.9
1750	1480.7	1479.2	1477.5	1475.6	1473.5	1471.2	1468.5	1465.3	1461.7	1457.4	1452.3	1446	1438.2
1800	1526.5	1524.9	1523.2	1521.3	1519.2	1516.7	1514	1510.8	1507.1	1502.7	1497.4	1491	1482.9
1850	1572.6	1571.1	1569.3	1567.4	1565.2	1562.7	1559.9	1556.6	1552.8	1548.3	1543	1536.4	1528.1
1900	1619.2	1617.6	1615.8	1613.8	1611.6	1609	1606.2	1602.8	1599	1594.4	1588.9	1582.2	1573.7
1950	1666	1664.4	1662.6	1660.6	1658.3	1655.7	1652.8	1649.4	1645.4	1640.7	1635.1	1628.3	1619.7
2000	1713.2	1711.5	1709.7	1707.6	1705.3	1702.7	1699.7	1696.2	1692.2	1687.4	1681.7	1674.7	1666
2050	1760.6	1758.9	1757	1754.9	1752.6	1749.9	1746.9	1743.4	1739.3	1734.4	1728.6	1721.5	1712.6
2100	1808.2	1806.5	1804.6	1802.5	1800.1	1797.4	1794.3	1790.8	1786.6	1781.7	1775.8	1768.6	1759.6
2150	1856.1	1854.4	1852.4	1850.3	1847.9	1845.1	1842	1838.4	1834.2	1829.2	1823.2	1815.9	1806.7
2200	1904.2	1902.4	1900.5	1898.3	1895.9	1893.1	1889.9	1886.3	1882	1877	1870.9	1863.5	1854.2

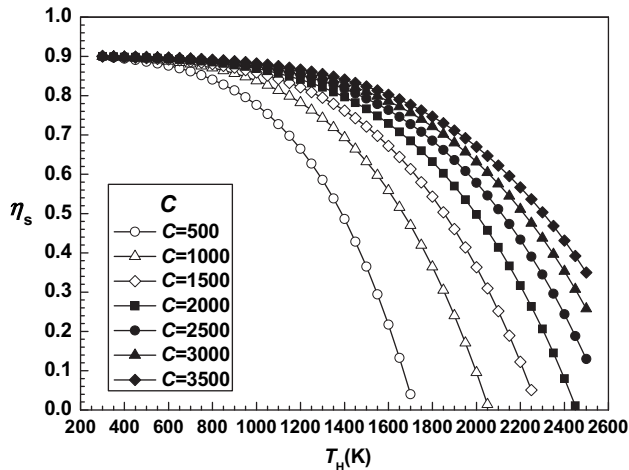


Fig. 4. Variation of the thermal efficiency of the collector for different the absorber temperature and the concentrating ratio.

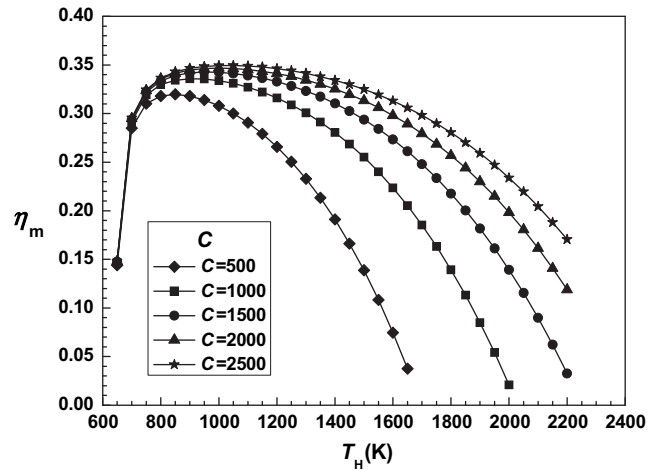


Fig. 6. Variation of the maximum power thermal efficiency of the dish system for different the absorber temperature and the concentrating ratio.

4. Numerical results and discussion

In order to evaluate the effect of the absorber temperature (T_H), the concentrating ratio (C), the effectiveness of the regenerator (ε_R) and the heat leak coefficient (k_0) on the solar-powered dish-Stirling heat engine system, all the other parameters will be kept constant as $x = 0.50$, $h_{HC} = h_{LC} = 200 \text{ W K}^{-1}$, $h_{HR} = 4 \times 10^{-8} \text{ W K}^{-4}$, $n = 1 \text{ mol}$, $\lambda = 2$, $R = 4.3 \text{ J mol}^{-1} \text{ K}^{-1}$, $C_v = 15 \text{ J mol}^{-1} \text{ K}^{-1}$, $\varepsilon = 0.90$, $k_0 = 2.5 \text{ W K}^{-1}$, $C = 1300$, $\varepsilon_R = 0.9$, $T_L = 320 \text{ K}$, $T_0 = 300 \text{ K}$, $h = 20 \text{ W m}^{-2} \text{ K}^{-1}$, $\delta = 5.67 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$, $T_H = 1100 \text{ K}$, $((1/M_1) + (1/M_2)) = 2.0 \times 10^{-5} \text{ s K}^{-1}$, $I = 1000 \text{ W m}^{-2}$. The results obtained are as follows.

4.1. Effect of T_H and C

The effect of the absorber temperature T_H and the concentrating ratio C on thermal efficiency of the collector and the Stirling engine as well as the system is shown in Figs. 4–7.

From Fig. 4, one can observe that the thermal efficiency of the collector decreases rapidly with increasing of the absorber temperature T_H , increases with the increasing of concentration ratio C . The maximum thermal efficiency is limited by the optical efficiency of the concentrator.

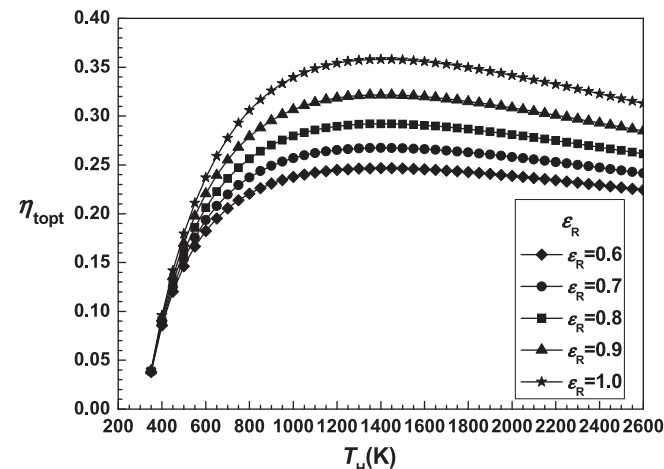


Fig. 5. Variation of the optimal thermal efficiency of the Stirling engine for different the absorber temperature and the effectiveness of the regenerator.

It can be seen from Fig. 5 that the optimal thermal efficiency of the Stirling engine increases rapidly at the beginning and decreases slowly afterward with the increasing of the absorber temperature, the optimal absorber temperature ranges from 1100 K to 1300 K, at which the thermal efficiency of the Stirling engine reaches to its maximum value. The cause is that conductive thermal bridging losses from the absorber to the heat sink affect the optimal thermal efficiency of the Stirling engine. It is also found that the optimal thermal efficiency of the Stirling engine increases with the increasing of the effectiveness of the regenerator and is influenced greatly by it.

From Figs. 6 and 7, it can be seen that for a given concentrating ratio, the maximum power thermal efficiency of the system increases with the increasing of the absorber temperature until the maximum thermal efficiency is reached and then decreases with the increasing of the absorber temperature; for a given absorber temperature, the maximum power thermal efficiency increases with the increasing of the concentrating ratio. The values of the optimum absorber temperature and the concentrating ratio are about 1100 K and 1300, respectively, which makes the thermal efficiency get up to its maximum value about 34% which is close to Carnot efficiency at about 50%, approximately. It is also found that for a given concentrating ratio, when the absorber temperature exceeds its optimum value, if keep increasing the absorber

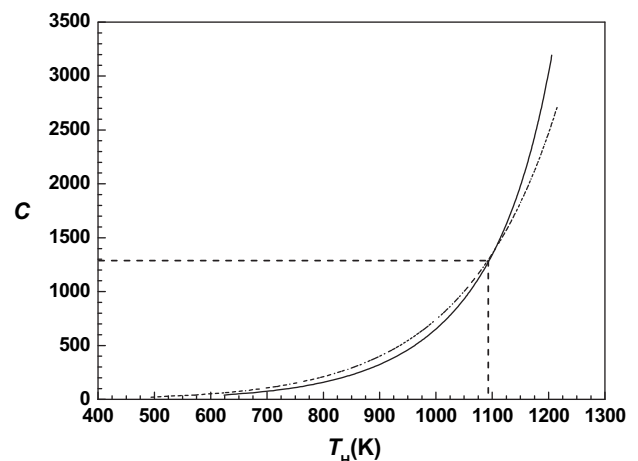


Fig. 7. The optimum absorber temperature and the concentrating ratio of the system.

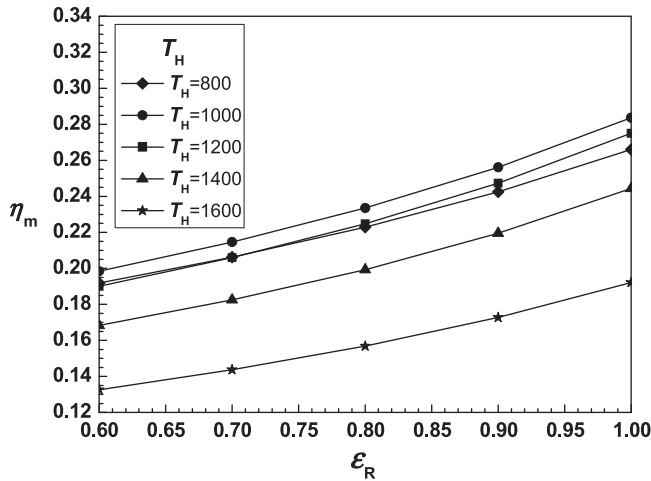


Fig. 8. Variation of the maximum power efficiency of the dish system for different the effectiveness of the regenerator and the absorber temperature.

temperature, the maximum power thermal efficiency of the system decreases rapidly. This shows that the range of the operation absorber temperature can not exceed its optimum temperature, which is very important for the solar dish collector because the absorber temperature varies with direct solar flux intensity and changes with time.

4.2. Effect of ε_R

The effect of the effectiveness of the regenerator on the maximum power thermal efficiency of the system is shown in Fig. 8. It is seen that the maximum power efficiency increases with the increasing of the effectiveness of the regenerator. Therefore, the most efficient and cost effective regenerator should be used for the Stirling engine.

4.3. Effect of k_0

The effect of the heat leak coefficient on the maximum power thermal efficiency of the system is shown in Fig. 9. It is seen from the figure that the heat leak coefficient reduces the maximum power efficiency of the system more at higher absorber temperature.

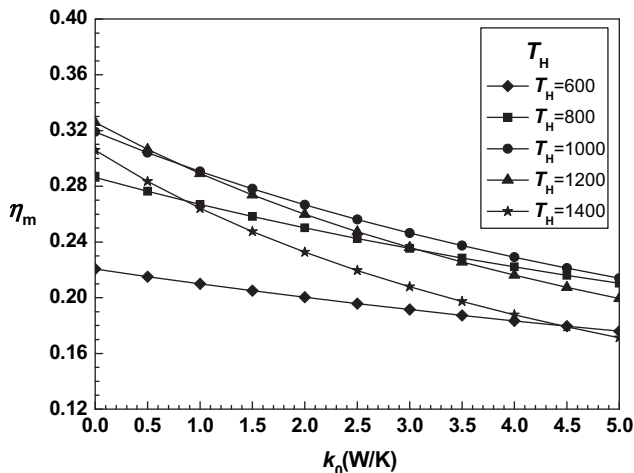


Fig. 9. Variation of the maximum power thermal efficiency of the dish system for different heat leak coefficient and the absorber temperature.

5. Conclusions

Finite-time thermodynamics has been applied to optimize the maximum power output and the corresponding thermal efficiency of the solar-powered dish-Stirling heat engine. Factors such as finite-rate heat transfer, regenerative heat losses, conductive thermal bridging losses and finite regeneration processes time are included in the analysis. It is found that the absorber temperature and the collector concentrating ratio as well as the effectiveness of regenerator are important characteristics of the system, which affect the maximum power thermal efficiency. The values of optimum absorber temperature and the collector concentrating ratio are about 1100 K and 1300, respectively. In order to achieve higher efficiency, it is desirable to use a high efficiency regenerator and to have as small as possible heat leak coefficient. Hence, the present analysis provides a new theoretical guidance for designing collectors and operating Stirling heat engines.

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